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A1

## DEMANDE DE BREVET D'INVENTION

(21)

**N° 75 03437**

(54) Procédé et dispositif d'équilibrage d'une pièce tournante.

(51) Classification internationale (Int. Cl.<sup>3</sup>). F 16 F 15/22; F 04 B 1/14.

(22) Date de dépôt ..... 4 février 1975, à 15 h 45 mn.

(33) (32) (31) Priorité revendiquée :

(41) Date de la mise à la disposition du  
public de la demande ..... B.O.P.I. - «Listes» n. 36 du 3-9-1976.

(71) Déposant : CREUSOT-LOIRE, résidant en France.

(72) Invention de :

(73) Titulaire : *Idem* (71)

(74) Mandataire :

L'invention a pour objet un procédé et un dispositif d'équilibrage d'une pièce articulée autour d'un centre d'oscillation et entraîné en rotation autour d'un axe passant par le centre d'oscillation.

5 L'invention s'applique en particulier à l'équilibrage de mécanismes à plateau oscillant tel que celui décrit dans la demande de brevet français 73-39888 déposé le 9 novembre 1973 par la même Société.

10 Le mécanisme décrit dans cette demande de brevet s'applique spécialement à un entraînement de pompes à pistons axiaux au moyen d'un plateau oblique tournant autour d'un centre d'oscillation par l'intermédiaire d'une articulation à cardan. Le plateau oscillant est entraîné en rotation par un maneton dirigé suivant l'axe passant par le centre de gravité du plateau et le centre  
15 d'oscillation et articulé sur une roue entraînée en rotation autour de l'axe de rotation.

Dans la même demande de brevet, on décrit un moyen de commande du débit de la pompe par déplacement radial de l'articulation du maneton sur la roue d'entraînement, ce déplacement commandant une variation de l'inclinaison de l'axe du maneton par rapport  
20 à l'axe de rotation.

Dans de telles pompes, le plateau oscillant tourne à une vitesse assez importante, et il est évidemment nécessaire de l'équilibrer.

25 L'équilibrage dynamique est difficile à réaliser du fait du réglage du débit qui se traduit par une variation de l'inclinaison de l'axe du plateau.

L'invention a pour objet un nouveau procédé d'équilibrage et un dispositif pour la mise en oeuvre de ce procédé, permettant  
30 de réaliser l'équilibrage quelle que soit l'inclinaison du maneton.

Conformément à l'invention, on dispose sur la roue d'entraînement deux masses d'équilibrage déplaçables radialement et dont les centres de gravité sont placés du côté opposé à l'axe du maneton par rapport à l'axe de rotation dans le plan passant par  
35 les dits axes, et l'on commande des déplacements radiaux différents des deux masses d'équilibrage en fonction de l'inclinaison de l'axe du maneton par rapport à l'axe de rotation, de telle sorte que la résultante des forces centrifuges exercées sur les deux masses constitue un balourd d'intensité et de position variables susceptible d'équilibrer le balourd exercé par la pièce tournante pour  
40

chaque inclinaison de celle-ci.

L'invention va maintenant être décrite en se référant à un mode de réalisation particulier d'un dispositif pour la mise en oeuvre du procédé selon l'invention, donné à titre d'exemple et  
5 représenté sur les dessins annexés.

La figure 1 est une vue partielle du mécanisme d'entraînement d'une pompe à pistons axiaux.

La figure 2 et la figure 3 sont des schémas illustrant le procédé d'équilibrage.

10 La figure 4 est une vue en perspective du dispositif d'équilibrage.

La figure 5 est une vue en bout du dispositif d'équilibrage.

La figure 6 est une vue en coupe suivant VI-VI, figure 5.  
15 Sur la figure 1, on a représenté le mécanisme d'entraînement d'une pompe à pistons axiaux comportant, à l'intérieur d'un carter 1, une pluralité de pistons 2 parallèles à un axe 3 et répartis régulièrement autour de ce dernier, la figure ne montrant en coupe que l'un des pistons 2. Les pistons sont reliés par des  
20 bielles 4 articulées à leurs extrémités à un plateau oblique 5 monté oscillant autour d'un centre d'oscillation 6. Dans le mode de réalisation représenté, le plateau 5 est monté tourillonnant autour d'un axe 51 sur une traverse 52 tourillonnant elle-même sur le carter 1 autour de tourillons 53 d'axes perpendiculaires à l'axe de  
25 rotation 3 et l'axe 51 de tourillonnement du plateau. L'ensemble constituant une articulation à cardan, on voit bien que le plateau 5 peut osciller dans toutes les directions autour du centre 6, point de rencontre des axes de tourillonnement 51 et 53.

D'autre part, le plateau est muni, du côté opposé au  
30 piston 2, d'un prolongement 54 qui se termine par un maneton 55 dont l'axe passe par le centre d'oscillation 6 et est perpendiculaire aux axes de tourillonnement 51 et 53. De préférence, le centre de gravité de l'ensemble est placé sur l'axe 55 du maneton.

Le maneton est relié par l'intermédiaire d'une articulation à rotule 57, à une roue 7 centrée sur l'axe de rotation 3 et entraînée en rotation autour du dit axe. Pour libérer la partie centrale de la roue, celle-ci est guidée le long de sa circonférence, dans l'extrémité 10 du carter 1, par deux roulements à rouleaux coniques 71.

40 Comme on le voit sur les figures 4 et 5, le palier à  
.../...

rotul s 57 du maneton est monté sur un coulisseau 72 guidé dans une glissière 73 ménagée dans la partie central de la roue 7 et dont l'axe passe par le centre de la roue 7 situé sur l'axe de rotation 3. Le coulisseau 72 est relié d'autre part à l'extrémité de la tige d'un petit vérin hydraulique monté sur la roue dans l'axe de coulissement, et alimenté par un dispositif d'asservissement de position 8, par l'intermédiaire d'un distributeur tournant, ce qui permet de régler, pendant la rotation de la roue, la position radiale du coulisseau 72, et ainsi l'inclinaison  $A$  de l'axe du maneton 550 par rapport à l'axe de rotation 3.

Comme on l'a indiqué dans la demande précédente déjà citée, l'ensemble qui vient d'être décrit permet de faire varier facilement et en marche le débit de la pompe puisque l'on peut régler l'inclinaison du plateau 5 de poussée des pistons et ainsi la course de ceux-ci.

Bien entendu, toutes les pièces mobiles seront équilibrées dans la mesure du possible, et c'est ainsi que le centre de gravité du plateau oscillant se trouvera normalement sur l'axe 550 du maneton. L'équilibrage de la roue d'entraînement 7 présente un peu plus de difficulté dans la mesure où le piston de commande du déplacement du coulisseau a une position radiale variable.

On pourra donc être obligé de tenir compte du balourd propre à la roue. Cependant, le balourd le plus important vient évidemment de la réaction du maneton sur la roue d'entraînement. Sur la figure 2, on a représenté schématiquement l'axe de rotation 3, le maneton 55 et son axe 550 passant par le centre d'oscillation  $O$  et sur lequel se trouve le centre de gravité  $G$  du plateau oscillant de masse  $M$ . Si  $d$  est la distance  $OG$  du centre de gravité au centre d'oscillation, la balourd provenant de la rotation du plateau oscillant a pour valeur :  $Md \sin A$ .

Les équations d'équilibre montrent que le balourd d'équilibrage doit se trouver dans le plan  $P$ , perpendiculaire à l'axe de rotation 3 et passant par le point d'intersection  $D$  de l'axe 3 avec la perpendiculaire à l'axe 550 élevée depuis le centre de l'articulation du maneton 55 sur la roue 7. D'habitude, on équilibre une masse tournante avec deux balourds se déplaçant dans deux plans d'équilibrage. L'une des caractéristiques de l'invention consiste précisément à n'utiliser qu'un seul plan d'équilibrage, en positionnant le centre de gravité  $G$  du plateau oscillant de telle manière que les réactions dynamiques au centre  $O$  d'oscillation soient nulles.

On voit que si l'on utilise une masse d'équilibrage de valeur  $M_e$ , celle-ci devra être placée dans le plan d'équilibrage et dans le plan passant par l'axe 3 et l'axe 550, à une distance radiale  $R_e$  de l'axe 3 telle que :

$$5 \quad M_e R_e = M_d \sin A$$

Il est bien évident que le balourd dépend de l'inclinaison  $A$  du plateau oscillant. Comme on ne peut pas faire varier la masse d'équilibrage  $M_e$ , on fera varier sa position de telle sorte que le rayon de giration du centre de gravité soit égal à :

$$10 \quad R_e = \frac{M_d \sin A}{M_e}$$

Cependant, compte tenu de ce qui précède, on voit qu'il faudra également faire varier la position du plan  $P_e$  d'équilibrage c'est à dire la distance  $e$  entre le plan  $P_e$  et le plan de la roue 7.

15 Selon l'invention, la variation du plan d'équilibrage est obtenue par l'utilisation de deux masses d'équilibrage placées à des distances radiales différentes de l'axe 3.

Pour plus de commodité, chaque masse d'équilibrage sera constituée de deux masselottes assujetties à tourner symétriquement autour de l'axe 3 de telle sorte que leur centre de gravité reste à une distance constante de l'axe. De la sorte, comme représenté sur la figure 3, si on appelle  $G_1$   $G_2$  les centres de gravité des masselottes,  $m_1$   $m_2$  décrivant ainsi un <sup>arc de</sup> cercle de rayon  $r$  autour du centre de la roue, on voit que les deux masselottes  $m_1$   $m_2$  sont équivalentes à une masse unique de valeur  $2m$  placée à une distance  $R$  du centre de rotation égale à la projection de  $r$  sur l'axe de symétrie.

Ainsi, en faisant tourner symétriquement deux masselottes autour du centre de la roue, on peut produire un balourd de valeur variable dans le plan perpendiculaire à l'axe de rotation et passant par les centres de gravité des deux masses.

Selon une caractéristique de l'invention, pour produire un balourd non seulement de valeur variable, mais dans un plan d'équilibrage de position variable le long de l'axe, on utilise deux balourds placés de part et d'autre du plan de la roue et que l'on appellera  $B$  et  $B'$ . Comme on le voit sur la figure 2, les deux balourds  $B$  et  $B'$  sont placés dans des plans d'équilibrage  $P$ ,  $P'$  qui se trouvent l'un,  $P$ , entre le centre d'oscillation  $O$  et le plan  $P_r$  de la roue, et l'autre,  $P'$ , de l'autre côté du plan  $P_r$  de la roue. La position des plans d'équilibrage  $P$  et  $P'$  dépend des conditions dans lesquelles on peut placer les masselottes sur la roue, mais cependant ./...

le plan  $P'$  devra être placé à une distance du plan  $P_r$  de la roue supérieure à la valeur maximale de la distance entre le plan  $P_r$  de la roue et la plan théorique d'équilibrage  $P_e$  pour l'inclinaison maximale du plateau oscillant.

5 Il sera possible, en faisant varier les valeurs des balourds  $B$  et  $B'$ , de s'arranger pour que la résultante se trouve dans le plan théorique d'équilibrage  $P_e$ .

Sur les figures 4, 5 et 6, on a représenté à titre d'exemple un dispositif permettant d'obtenir ce résultat.

10 On a déjà indiqué que la roue 7 est guidée et entraînée par sa périphérie, ce qui permet de libérer entièrement la partie centrale dans laquelle se trouvent essentiellement le coulisseau 72 de support de l'articulation du maneton et le vérin 73 de déplacement du coulisseau.

15 La roue 7 est munie dans son plan équatorial d'un voile 75 dans la partie dentrale duquel est ménagée une ouverture radiale 750 à l'intérieur de laquelle peut se déplacer le coulisseau. Les guides 73 du coulisseau sont placés le long des côtés de l'ouverture 750, parallèles à l'axe 720 de déplacement du coulisseau. De part et d'autre de la cloison 75, sont placées deux paires de masselottes 20 8 et 8'. Chaque paire se compose de deux masselottes guidées symétriquement de part et d'autre de l'axe 720 le long de glissières circulaires centrées sur l'axe de rotation 3. Comme on le voit plus en détail sur la figure 6, chaque masselotte 8 est guidée entre une 25 glissière intérieure 76 ayant la forme d'un secteur circulaire solidaire de la glissière correspondante 73 du coulisseau et une glissière extérieure 77 fixée sur une nervure solidaire de la périphérie de la roue 7. Chaque masselotte 8 est pourvue de congés s'adaptant aux deux glissières et s'appuie d'autre part sur une nervure 30 circulaire 741 ménagée sur la paroi 75. Ainsi, chaque masselotte peut être déplacée le long des glissières 77 et 76, son centre de gravité restant à une distance constante de l'axe 3 dans un plan perpendiculaire audit axe.

Les glissières circulaires sont placées de telle sorte que 35 les masselottes puissent se rapprocher le plus possible de l'axe 720, comme on l'a représenté sur la partie droite de la figure 5, et s'écarter dudit axe jusqu'à ce que leur centre de gravité vienne dans le plan radial perpendiculaire à l'axe 720, dans la position représentée sur la partie gauche de la figure 5. On voit bien qu'40 ainsi, le balourd résultant de l'action des deux masselottes varie

entre une valeur nulle et une valeur maximale. Selon la forme et les poids donnés aux masselottes, on peut choisir les plans P et P' d'équilibre et les valeurs maximales des balourds.

Sur la figure 2, on a représenté les plans P et P' dans lesquels sont produits les balourds B et B'. Ceux-ci se trouvent respectivement à des distances a et b du plan de la roue dans lequel s'exerce éventuellement le balourd propre B<sub>r</sub> de la roue. Si l'on prend comme référence l'axe Oz perpendiculaire à l'axe de rotation 3 et passant par le centre de l'articulation du maneton, l'équation de projection sur l'axe Oz donne :

$$(3) B' + B = B_e + B_r$$

Es étant le balourd d'équilibre qui doit être produit dans le plan théorique P<sub>e</sub>.

L'équation de moment par rapport au centre O nous donne :

$$(4) a B' - b B = e B_e$$

On en tire les valeurs qu'il faut donner au balourd produit par les masselottes :

$$(5) B' = \frac{(e + b) B_e + b B_r}{a + b} \text{ et :}$$

$$(6) B = B_e + B_r - B'$$

Par ailleurs, si on appelle l la distance OC entre le centre d'oscillation et le centre de la roue, on peut écrire :

$$(7) e = l \operatorname{tg}^2 A.$$

Pour toute valeur de l'inclinaison A du plateau oscillant la relation (1) donne B<sub>e</sub> = M<sub>e</sub> R<sub>e</sub> = Md sin A. Si l'on s'est fixé à l'avance les distances a et b des plans d'équilibre P et P' par rapport au plan de la roue, les relations 5 et 6 permettant de calculer les balourds extérieur et intérieur B' et B.

Si m et m' sont les masses des masselottes d'équilibre, r et r' les rayons de giration des centres de gravité des masselottes, R et R' les projections sur l'axe 720 des positions des centres des masselottes, et L l'inclinaison par rapport à l'axe 720 des rayons passant par les centres de gravité, sachant que :

$$B = 2m R = 2m r \cos L \text{ et :}$$

$$B' = 2m' R' = 2m' r' \cos L,$$

il est facile de déduire des valeurs des balourds B et B' calculées précédemment, l'angle L en fonction de l'angle A.

Il suffit alors de mettre au point un système mécanique par lequel la position angulaire des masselottes sera liée à la position radiale du coulisseau 72 de telle sorte qu'à chaque valeur



de A corresponde la valeur désirée d L. Il ne s'agit alors que d'un problème de choix d'un mécanisme qui peut être résolu plus ou moins empiriquement.

On a trouvé que le dispositif mécanique représenté à  
 5 titre d'exemple sur le figures permettait d'obtenir cette variation de l'inclinaison des masselottes.

Comme on l'a représenté sur les figures 4, 5 et 6, ce dispositif mécanique se compose essentiellement, pour chaque paire de masselottes d'une paire de crémaillères 91, articulées sur le  
 10 coulisseau autour d'axes parallèles à l'axe de rotation, et s'étendant symétriquement de part et d'autre de l'axe 720. Chaque crémaillère est assujettie au moyen d'un étrier 92 à engrener sur un pignon 93 monté rotatif sur la paroi 75 autour d'un pivot excentré 94. Les crémaillères 92 sont elles-mêmes solidaires en rotation  
 15 des pignons excentrés 93.

Sur chaque pivot 94 est centré un pignon 95 qui engrène sur un secteur denté 85 solidaire de la masselotte 8 et centré sur l'axe de rotation 3.

Ainsi, le déplacement du coulisseau 72 au moyen du vérin  
 20 73 entraîne, par l'intermédiaire des crémaillères 91, la rotation des pignons excentrés 93 et par conséquent des pignons 95, ceux-ci commandant à leur tour la rotation des masselottes autour de l'axe 3 puisque leurs pivots 94 sont fixes par rapport à la roue. Compte tenu de l'excentrement des pignons 93 et du rapport de démultipli-  
 25 cation entre les pignons 95 et les secteurs 85, on voit que le déplacement du coulisseau, c'est à dire la variation de l'angle A entraîne une variation irrégulière de l'angle L.

On voit qu'il est possible de mettre le tout en équation, et, en faisant varier certains paramètres dans les limites permises  
 30 par les considérations mécaniques de construction (encombrement, résistance etc ...) on peut trouver, par exemple au moyen d'un calculateur, la solution optimale permettant de suivre au mieux la variation théorique de L en fonction de A établie comme indiqué précédemment.

Bien entendu, pour simplifier les calculs, on pourrait ne  
 35 pas tenir compte du balourd de la roue venant de la variation du piston de commande du coulisseau, et équilibrer la roue une fois pour toutes.

Inversement, pour obtenir un résultat plus exact, on  
 40 pourrait tenir compte dans le balourd de la roue de l'hétérocinétisme

de l'articulation à cardan par l'intermédiaire de laquelle le plateau est monté oscillant et qui nécessite des balourds secondaires. Cependant, on considérera généralement cet effet comme négligeable devant le balourd principal.

- 5 D'autre part, le système mécanique de variation de l'angle  $L$  en fonction de la position du maneton n'a été donné évidemment qu'à titre indicatif, et on pourrait imaginer d'autres systèmes qui ne seraient d'ailleurs pas obligatoirement mécaniques, mais par exemple hydrauliques.

REVENDEICATIONS

1. - Procédé d'équilibrage d'une pièce articulée autour d'un centre d'oscillation et entraînée en rotation autour d'un axe passant par le centre d'oscillation par l'intermédiaire d'un maneton dirigé suivant l'axe passant par le centre de gravité de la pièce et le centre d'oscillation et relié à une roue d'entraînement tournant autour de l'axe de rotation, par l'intermédiaire d'une articulation déplaçable radialement sur la roue, le déplacement radial de l'articulation commandant une variation de l'inclinaison de l'axe du maneton par rapport à l'axe de rotation, caractérisé par le fait que l'on dispose sur la roue d'entraînement deux masses d'équilibrage déplaçables radialement et dont les centres de gravité sont placés du côté opposé à l'axe du maneton par rapport à l'axe de rotation, dans le plan passant par lesdits axes, et que l'on détermine des écartements radiaux différents des deux masses d'équilibrage en fonction de l'inclinaison de l'axe du maneton par rapport à l'axe de rotation de telle sorte que la résultante des forces centrifuges exercées par les deux masses constitue un balourd d'intensité et de position variables, susceptible d'équilibrer le balourd exercé par la pièce tournante pour chaque inclinaison de celle-ci.

2. - Procédé d'équilibrage selon la revendication 1, caractérisé par le fait que la position des centres de gravité des masses d'équilibrage est réglée pour chaque inclinaison de l'axe du maneton de telle sorte que la résultante des forces centrifuges exercées sur les deux masses passe par le point d'intersection avec l'axe de rotation de la perpendiculaire à l'axe du maneton passant par le centre de l'articulation du maneton sur la roue.

3. - Dispositif d'équilibrage d'une pièce articulée autour d'un centre d'oscillation et portant un maneton centré sur l'axe passant par le centre de gravité de la pièce et par le centre d'oscillation ledit maneton étant articulé sur une roue entraînée en rotation autour d'un axe passant par le centre d'oscillation et munie d'un moyen de variation de l'inclinaison de la pièce par rapport à l'axe de rotation par déplacement radial de l'articulation du maneton sur la roue d'entraînement, caractérisé par le fait qu'il comprend deux masses d'équilibrage solidaires en rotation de la roue d'entraînement et dont les centres de gravité sont placés de l'autre côté du maneton par rapport à l'axe de rotation dans le plan passant par lesdits axes, et à des distances différentes du centre d'oscillation, et un moyen de commande d'écartements radiaux différents

des centres de gravité des deux masses par rapport à l'axe de rotation en fonction de l'inclinaison de l'axe.

4. - Dispositif d'équilibrage d'une pièce articulée selon la revendication 3, caractérisé par le fait que chaque masse d'équilibrage est constituée par deux masselottes solidaires en rotation de la roue d'entraînement et dont les centres de gravité sont déplaçables symétriquement par rotation autour de l'axe de rotation de la roue de part et d'autre du plan passant par l'axe de rotation et l'axe du maneton et que le moyen de commande des écartements radiaux des masses d'équilibrage détermine, pour chaque masse d'équilibrage, des rotations symétriques des masselottes correspondantes autour de l'axe de rotation.

5. - Dispositif d'équilibrage d'une pièce articulée, selon la revendication 4, dans laquelle l'articulation du maneton est montée sur une pièce coulissante radialement sur la roue, caractérisé par le fait que le moyen de commande des écartements radiaux du centre de gravité de chaque masse d'équilibrage comprend, pour chaque masselotte, une crémaillère montée pivotante sur la pièce coulissante et en prise avec un premier pignon denté tournant autour d'un axe excentré sur lequel est centré et calé un second pignon engrenant sur un secteur denté ménagé sur la masselotte et centré sur l'axe de rotation, chaque masselotte étant guidée le long d'une glissière circulaire ménagée sur la roue et centrée sur l'axe de rotation.

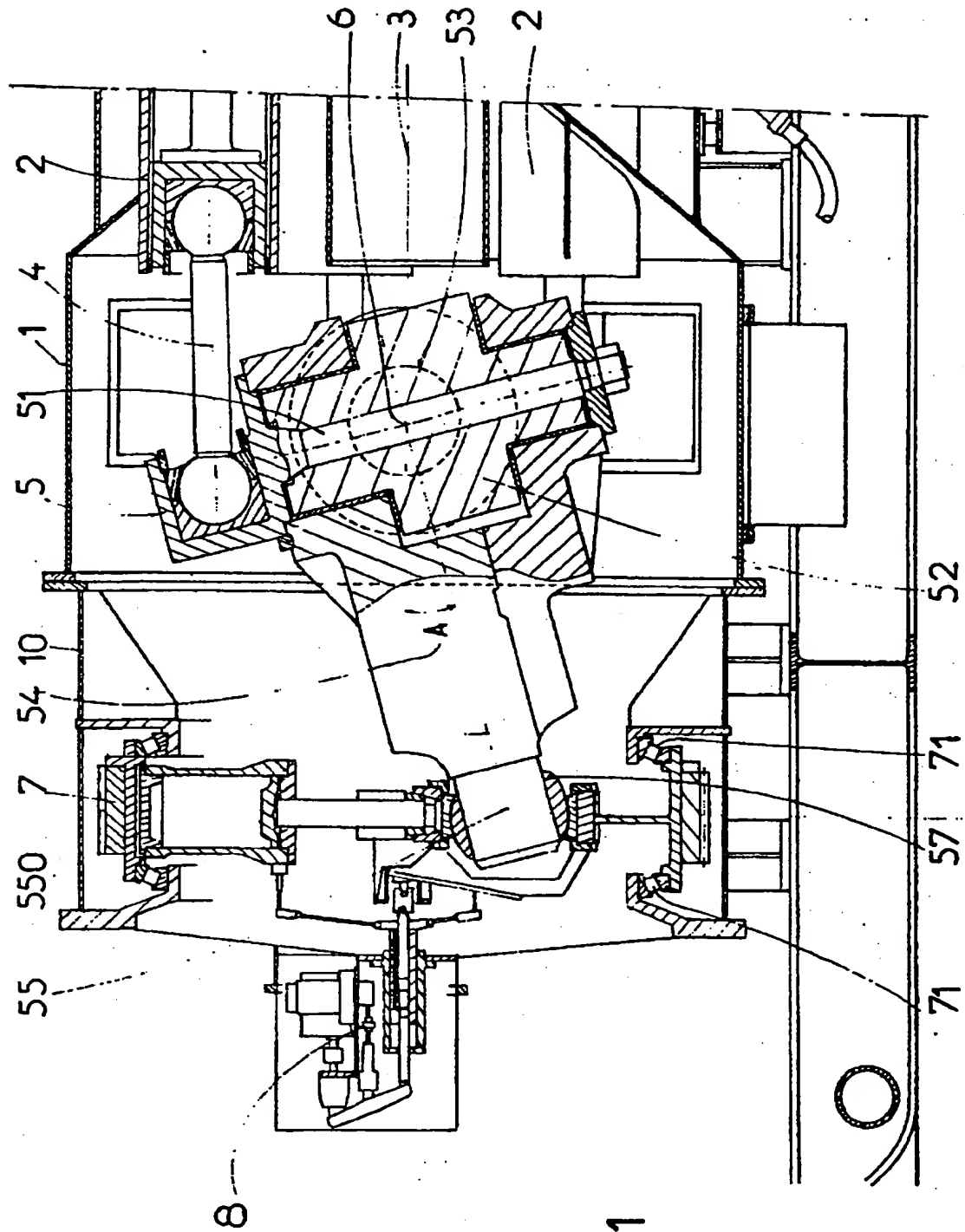


FIG 3

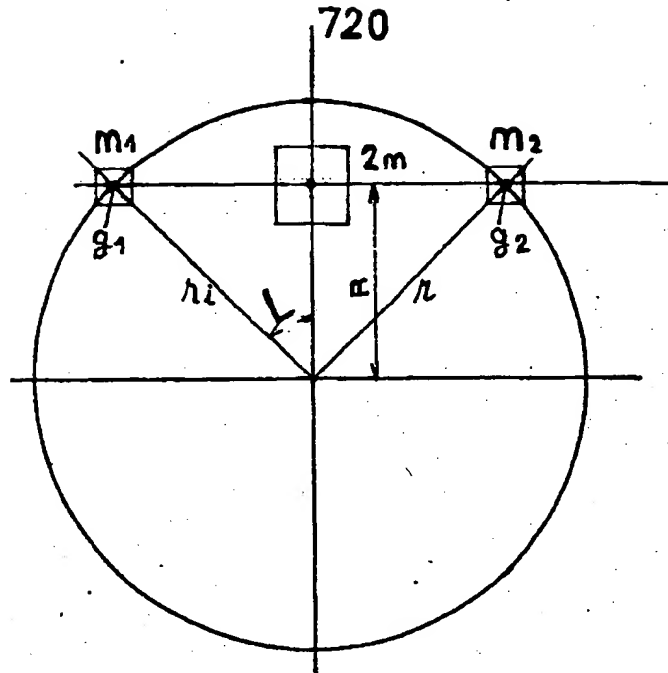
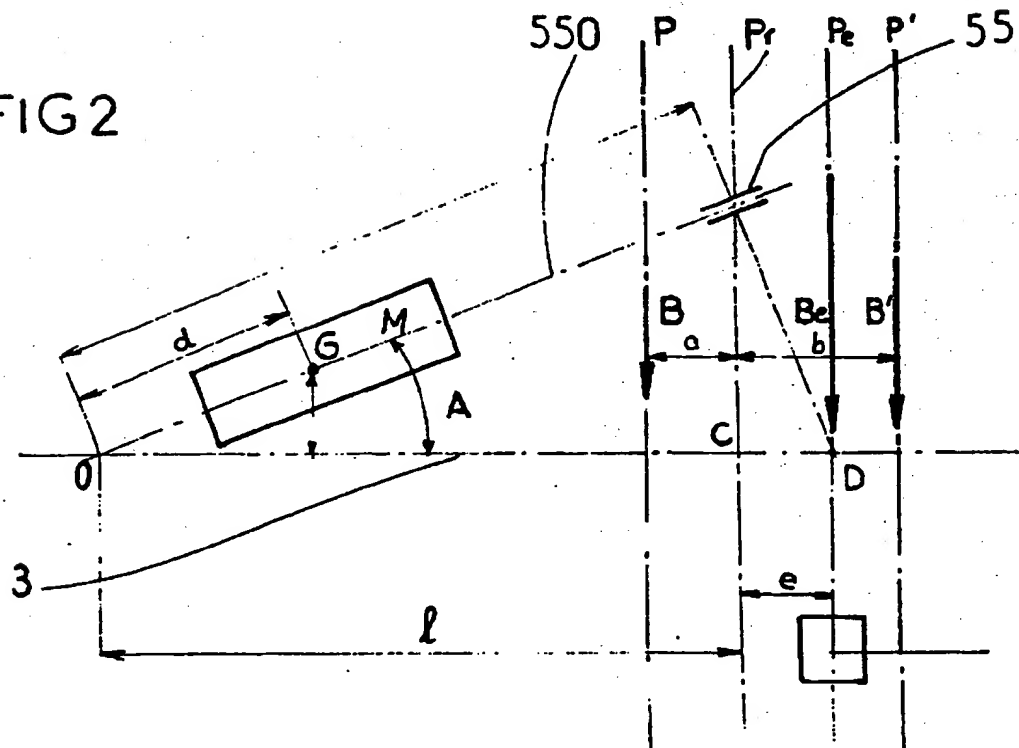


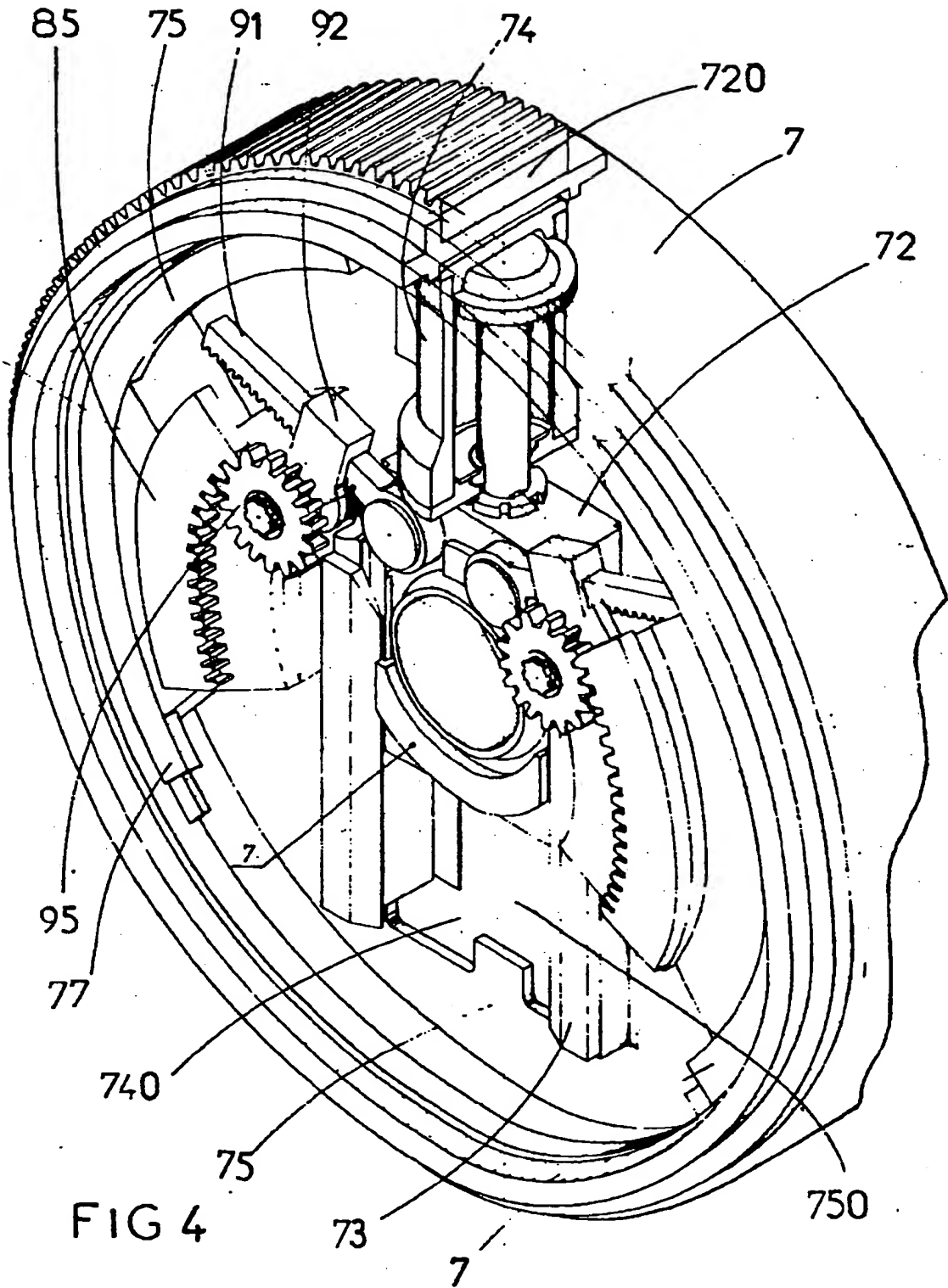
FIG 2

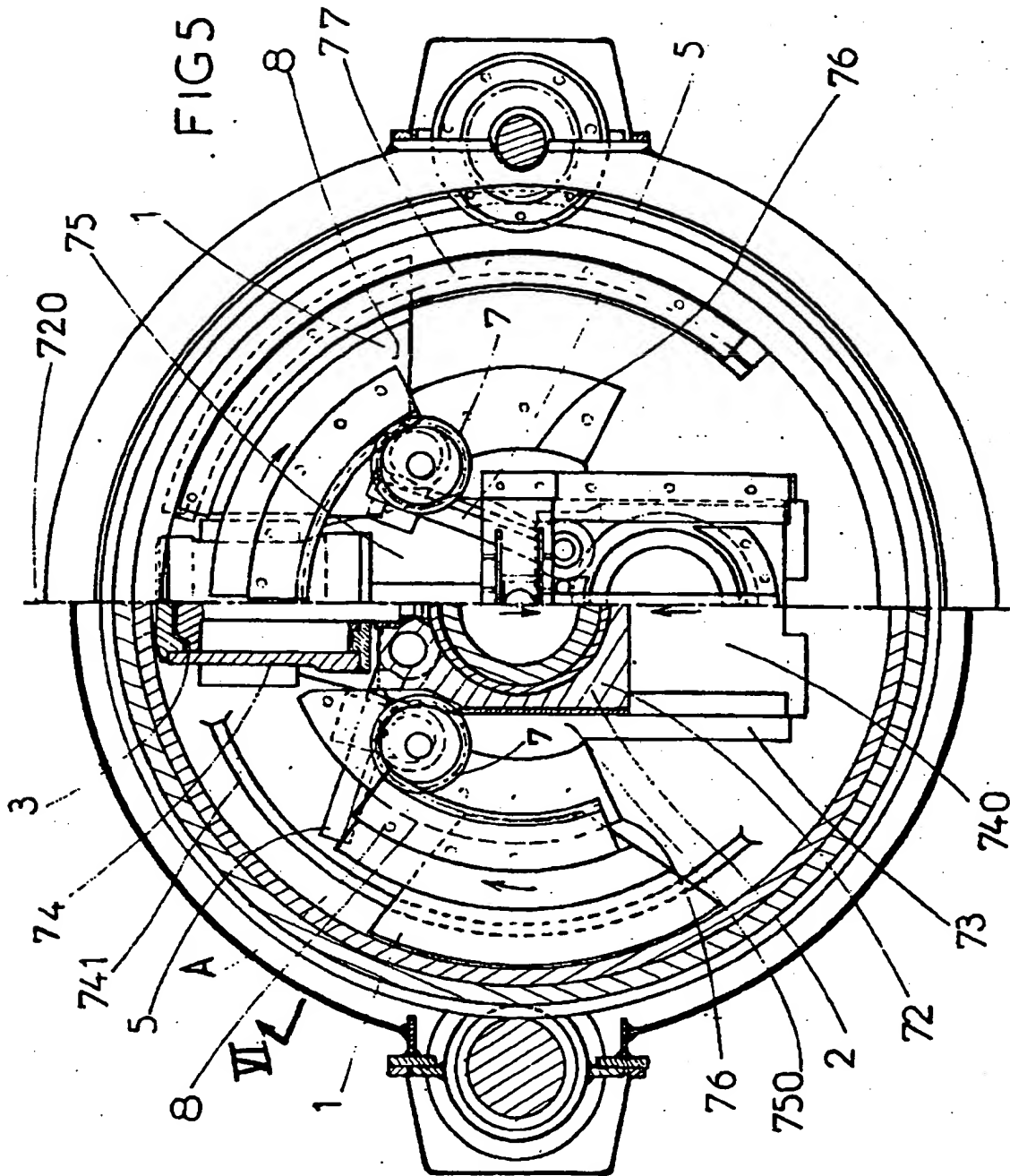


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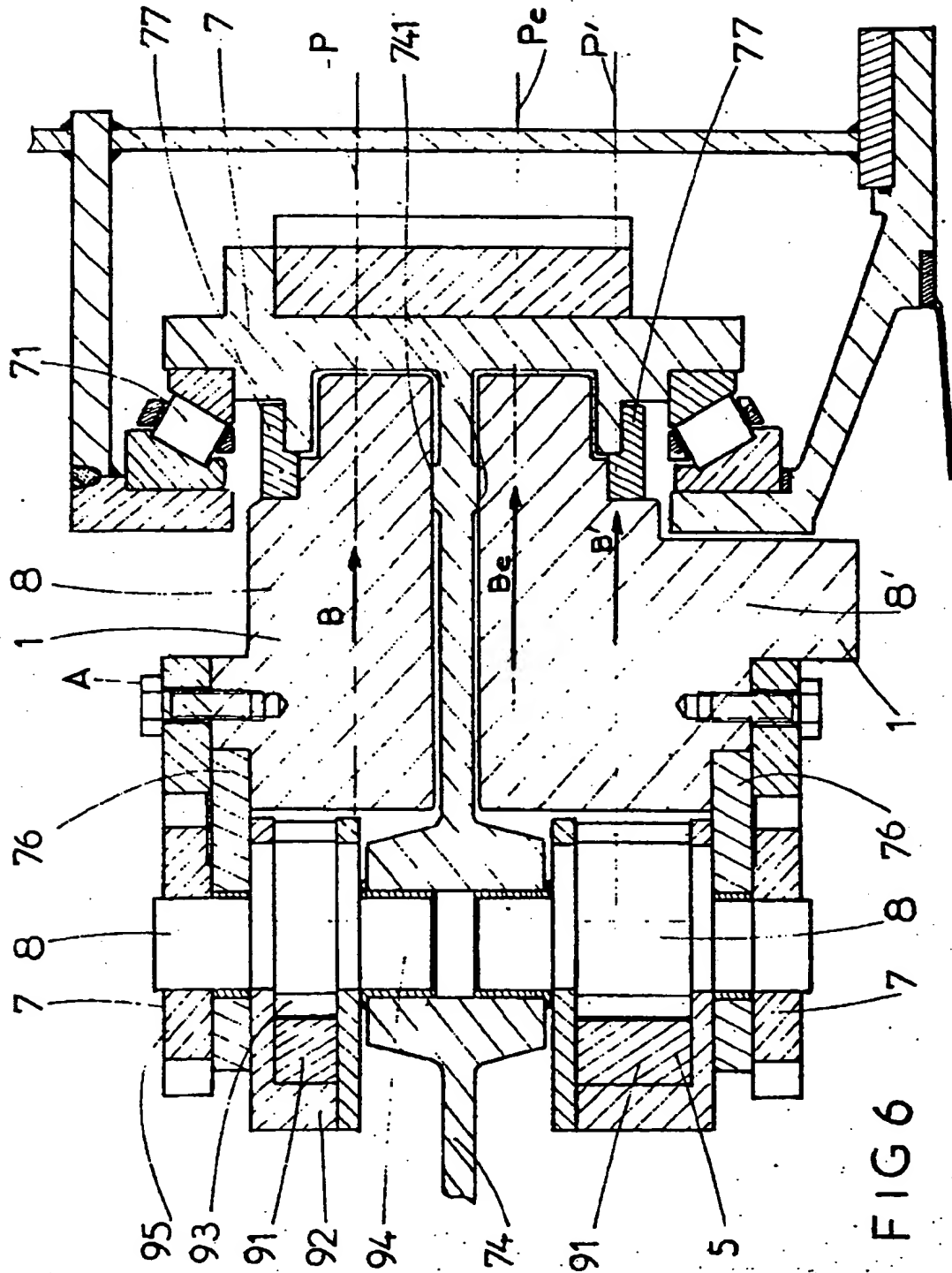


FIG 6

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**PATENT APPLICATION**

(21)

**No. 75 03437**

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(54) Process and device for balancing a rotary part

(51) International Classification No.<sup>2</sup>: F 16 F 15/22; F 04 B 1/14.

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This invention concerns a process and a device for balancing a part that pivots around a center of oscillation, rotationally driven around an axis passing through the center of oscillation.

In particular, the invention is applicable to the balancing of swash plate mechanisms such as the one described in French Patent Application 73-39888 filed on November 9, 1973 by the same company.

The mechanism described in that patent application is especially applicable to an axial piston pump driven by means of a diagonal plate turning around a center of oscillation through the intermediary of a universal joint. The swash plate is rotationally driven by a crank pin running along the axis passing through the center of gravity of the plate and the center of oscillation, and pivoting around a wheel rotationally driven around the rotation axis.

The same patent application describes a means of controlling the output of the pump by the radial movement of the articulation of the crank pin on the drive wheel, with said movement controlling a variation in the pitch of the axis of the crank pin relative to the center of rotation axis.

In such pumps, the swash plate turns at a fairly high speed, and it is therefore necessary to balance it.

Dynamic balancing is difficult to achieve due to the output regulation, which varies the pitch of the axis of the plate.

The object of the invention is a ~~new balancing process and a device to implement this process that enables balancing regardless of the pitch of the crank pin.~~

According to the invention, two radially movable balance weights are placed on the drive wheel, and their centers of gravity are placed on the side the axis of the crank pin opposite from the rotation axis, in the plane passing through said axes, and the different radial motions of the two balance weights are controlled according to the pitch of the axis of the crank pin relative to the rotation axis, such that the resultant of the centrifugal forces exerted on the two weights constitutes a counterweight whose variable intensity and position are capable of balancing the counterweight exerted by the rotary part for every possible pitch.

The invention will now be described with reference to a particular embodiment of a device for implementing the process according to the invention, provided as an example and illustrated in the attached drawings.

Figure 1 is a partial view of the drive mechanism of an axial piston pump.

Figure 2 and Figure 3 are diagrams illustrating the balancing process.

Figure 4 is a perspective view of the balancing device.

Figure 5 is an end view of the balancing device.

Figure 6 is a cross section along VI-VI from Figure 5.

Figure 1 illustrates the drive mechanism of an axial piston pump comprising, inside a housing 1, ~~a plurality of pistons 2 parallel to an axis 3, and evenly distributed around the latter.~~ This figure shows a cross section of only one of the pistons 2. The pistons are connected by connecting rods 4, articulated at their ends to a diagonal plate 5, oscillatably mounted around a center of oscillation 6. In the illustrated embodiment, the plate 5 is swivelably mounted round an axis 51 on a cross-member 52, itself swivelably mounted on the housing 1, around trunnions 53 whose axes are perpendicular to the rotation axis 3 and the trunnion axis 51 of the plate. Since this assembly constitutes a ~~universal joint,~~ it is clear that the plate 5 is able to oscillate in all directions around the center 6, the point where the trunnion axes 51 and 53 meet.

Furthermore, on the side opposite the piston 1, the plate is equipped with an extension 54 that ends in a crank pin 55 whose axis passes through the center of oscillation 6 and is perpendicular to the trunnion axes 51 and 53. Preferably, the center of gravity of the assembly is placed on the axis 550 of the crank pin.

The crank pin is connected, through the intermediary of a ball and socket joint 57, to a wheel 7 centered on the rotation axis 3 and rotationally driven around said axis. To liberate the central part of the wheel, it is guided, along its circumference, in the end 10 of the housing 1, by two tapered roller bearings 71.

As shown in Figures 4 and 5, the spherical bearing 57 of the crank pin is mounted on a sliding nut 72 guided in a guideway 73 in the central part of the wheel 7, whose axis passes through the center of the wheel 7 located on the rotation axis 3.

As indicated in the aforementioned previous application, the assembly that has just been described makes it possible to easily vary the output of the pump while it is operating, since the pitch of the piston thrust plate 5, and thus its travel, can be adjusted.

Of course, all mobile parts are balanced, to the extent possible, and, thus, the center of gravity of the oscillating plate is normally on the axis 550 of the crank pin. Balancing the drive wheel 7 is somewhat more difficult, inasmuch as the piston that controls the travel of the sliding nut has a radially variable position.

Therefore, it may be necessary to take the counterweighting of the wheel itself into account. However, the greatest counterweighting clearly comes from the reaction of the crank pin on the drive wheel. Figure 2 schematically illustrates the rotation axis 3, the crank pin 55 and its axis 550 passing through the center of oscillation 0, on which the center of gravity G of the oscillating plate with a mass M is located. If d is the distance OG from the center of gravity to the center of oscillation, the counterweighting from the rotation of the oscillating plate is equal to  $M_d \sin A$ .

The balance equations show that the balancing counterweight should be in the plane  $P_e$  perpendicular to the rotation axis 3, passing through the point of intersection D of the axis 3, with the perpendicular to the axis 550 raised from the center of articulation of the crank pin 55 on the wheel 7. A turning weight is generally balanced with two counterweights traveling in two balancing planes. One of the characteristics of the invention consists precisely of using only one balancing plane, by positioning the center of gravity G of the oscillating plane such that the dynamic reactions at the center of oscillation O are nil.

We see that, if we use a balancing weight with a value  $M_e$ , it should be placed in the balancing plane and in the plane passing through the axis 3 and the axis 550, at a radial distance  $R_e$  from the axis 3, such that:

$$M_e R_e = M_d \sin A$$

It is clear that the counterweighting depends on the pitch A of the oscillating plate. Since the balancing weight  $M_e$  cannot be varied, its position is varied such that the radius of turn of the center of gravity is equal to:

$$R_e = \frac{M_d \sin A}{M_e}$$

However, in view of the foregoing, we see that it is also necessary to vary the position of the balancing plane  $P_e$ , i.e., the distance e between the plane  $P_e$  and the plane of the wheel 7.

According to the invention, the variation of the balancing plane is achieved by using two balancing weights placed at different radial spacings from the axis 3.

For greater convenience, each balancing weight is composed of two centrifugal weights affixed so as to turn symmetrically around the axis 3 such that their center of gravity remains at a constant distance from the axis. In that way, as illustrated in Figure 3, if we call  $g_1$  and  $g_2$  the centers of gravity of the centrifugal weights, with  $m_1$  and  $m_2$  arc with thus describing an/ circle with a radius r around the center of the wheel, we see

that the two centrifugal weights  $m_1$  and  $m_2$  are equivalent to a single weight with a value of  $2m$  placed at a distance R from the center of rotation equal to the projection of r on the axis of symmetry.

Thus, by causing the two centrifugal weights to turn symmetrically around the center of the wheel, we can produce a counterweight with a variable value in the plane perpendicular to the rotation axis, passing through the centers of gravity of the two weights.

According to one characteristic of the invention, to produce a counterweight not only with a variable value, but in a balancing plane with a variable position along the axis, we use two counterweights placed on either side of the plane of the wheel, which we

call B and B'. As illustrated in Figure 2, the two counterweights B and B' are placed in balancing planes P and P', one of which, P, is located between the center of oscillation O and the plane P<sub>r</sub> of the wheel, and the other, P', on the other side of the plane P<sub>r</sub> of the wheel. The position of the balancing planes P and P' depends on the conditions under which the centrifugal weights may be placed on the wheel. However, the plane P' must be placed at a distance from the plane P<sub>r</sub> of the wheel that is greater than the maximum value of the distance e between the plane P<sub>r</sub> of the wheel and the theoretical balancing plane P<sub>e</sub> for the maximum pitch of the oscillating plate.

By varying the values of the counterweights B and B', it is possible to arrange for the resultant to be located in the theoretical balancing plane P<sub>e</sub>.

Figures 4, 5, and 6 illustrate, as an example, a device that provides this result.

We have already indicated that the wheel 7 is guided and driven by its periphery, which makes it possible to entirely liberate the central part where the sliding nut 72 supporting the hinge of the crank pin and the travel cylinder 73 of the sliding nut are essentially located.

The wheel 7 is equipped, in its equatorial plane, with a disc 75 in whose central part a radial opening 750 is pierced, within which the sliding nut can travel. The guides 73 of the sliding nut are placed along the sides of the opening 750, parallel to the travel axis 720 of the sliding nut. On either side of the baffle 75, there are two pairs of centrifugal weights 8 and 8'. Each pair is composed of two centrifugal weights symmetrically guided on either side of the axis 720 along circular guideways centered on the rotation axis 3. As illustrated in greater detail in Figure 6, each centrifugal weight 8 is guided between an inner guideway 76 in the shape of a sector of a circle, which is integral with the corresponding guideway 73 of the sliding nut, and an outer guideway 77 affixed to a bead that is integral with the periphery of the wheel 7. Each centrifugal weight 8 is equipped with fillets that adapt to the two guideways and rest also on a circular bead 741 on the wall 75. Thus, each centrifugal weight may be moved along the guideways 77 and 76, with its center of gravity remaining at a constant distance from the axis 3 in a plane perpendicular to said axis.

The circular guideways are placed such that the centrifugal weights may come as close as possible to the axis 720, as illustrated at the right in Figure 5, and may move away from said axis until their center of gravity comes into the radial plane perpendicular to the axis 720, in the position illustrated at the left in Figure 5. Thus, we see that the counterweight resulting from the action of the two centrifugal weights varies between a nil value and a maximum value. Depending on the shape and mass of the centrifugal weights, we may choose the balancing planes P and P' and the maximal values of the counterweights.

Figure 2 illustrates the planes P and P' in which the counterweights B and B' are produced. These are located, respectively, at distances a and b from the plane of the wheel in which the counterbalance  $B_r$  of the wheel itself may be located. If we take as a reference the axis Oz perpendicular to the rotation axis 3, passing through the center of articulation of the crank pin, the equation for the projection on the axis Oz gives:

$$(3) B' + B - B_e + B_r$$

$B_e$  being the balancing counterweight that must be produced in the theoretical plane  $P_e$ .

The equation for the momentum relative to the center O gives:

$$(4) a B' - b B = e B_e$$

From this, we derive the values that must be used for the counterweight produced by the centrifugal weights:

$$(5) B' = \frac{(e+b) B_e + b B_r}{a+b}$$

and:  $(6) B = B_e + B_r - B'$

Furthermore, if we call l the distance OC between the center of oscillation and the center of the wheel, we may write:

$$(7) e = l \lg^2 A$$

For any pitch value A of the oscillating plate, the relationship (1) gives  $B_e = M_e R_e = M_d \sin A$ . If we determine in advance the distances a and b of the balancing planes P and P' relative to the plane of the wheel, relationships 5 and 6 make it possible to calculate the outer and inner counterweights B' and B.

If m and m' are the masses of the balancing centrifugal weights, r and r' the radii of turn of the centers of gravity of the centrifugal weights, R and R' the projections on the axis 720 of the positions of the centers of the centrifugal weights, and L the pitch relative to the axis 720 of the radii passing through the centers of gravity, where:

$$B = 2m R = 2m r \cos L$$

and:  $B' = 2m' R' = 2m' r' \cos L$ ,

it is easy to deduce from the previously calculated values of the counterweights B and B' the angle L as a function of the angle A.

Then, it is sufficient to develop a mechanical system whereby the angular position of the centrifugal weights is linked to the radial position of the sliding nut 72 such that the desired value L corresponds to each value of A. Then, there is only the problem of choosing a mechanism, which can be determined more or less empirically.

We have found that the mechanical device illustrated as an example in the figures made it possible to achieve this variation in the pitch of the centrifugal weights.

As illustrated in Figures 4, 5, and 6, for each pair of centrifugal weights, this mechanical device is essentially composed of a pair of racks 91, articulated around the sliding nut around axes parallel to the rotation axis, running symmetrically along either

side of the axis 720. Each rack is affixed, by means of a U bolt 92 that meshes with a pinion 93 rotationally mounted on the wall 75 around an off-center pivot 94. The racks 92 themselves are rotationally integral with the off-center pivots 93.

On each pivot 94 is centered a pinion 95 that meshes with a toothed sector that is integral with the centrifugal weight 8 and centered on the rotation axis 3.

Thus, the travel of the sliding nut 72 by means of the cylinder 73 drives, through the intermediary of the racks 91, the rotation of the off-center pivots 93 and as a result the pivots 95, which in turn control the rotation of the centrifugal weights around the axis 3, since their pivots 94 are stationary relative to the wheel. In view of the off-centering of the pinions 93 and the gear ratio between the pinions 95 and the sectors 85, we see that the travel of the sliding nut, i.e., the variation of the angle  $A$ , results in an irregular variation in the angle  $L$ .

We see that it is possible to set up equations for everything and, by varying certain parameters within the limits allowed by the mechanical construction considerations (dimensions, strength, etc.), we can find, using a computer for example, the optimal solution for best following the theoretical variation in  $L$  as a function of  $A$ , established as indicated above.

Of course, to simplify the calculations, it is possible to not take into account the counterweighting of the wheel due to the variation in the control piston of the sliding nut, and to balance the wheel once and for all.

Conversely, to achieve the most accurate results, it is possible to take into account the counterweighting of the wheel due to the non-colinear motion of the universal joint through the intermediary of which the plate is oscillatingly mounted, which requires secondary counterweights. However, this effect may be generally considered negligible relative to the primary counterweight.

Furthermore, the mechanical system for varying the angle  $L$  as a function of the position of the crank pin is provided solely for information and it is possible to imagine other systems that are not necessarily mechanical, but rather hydraulic, for example.



### CLAIMS

1. Process for balancing a rotary part articulated around a center of oscillation and rotationally driven around an axis passing through the center of oscillation through the intermediary of a crank pin running along the axis passing through the center of gravity of the part and the center of oscillation and connected to a drive wheel turning around the rotation axis, through the intermediary of a radially movable articulation on the wheel controlling the variation of the pitch of the axis of the crank pin relative to the rotation axis, wherein, placed on the drive wheel, are two radially movable balancing weights whose centers of gravity are placed on the side of the axis of the crank pin opposite from the rotation axis, in the plan passing through said axes, and wherein the different radial spacings of the two balancing weights are determined by the pitch of the axis of the crank pin relative to the rotation axis, such that the resultant of the centrifugal forces exerted by the two weights constitutes a counterweight with variable intensity and position, capable of balancing the counterweighting exerted by the turning part for every possible pitch.

2. Balancing process according to Claim 1, wherein the position of the centers of gravity of the balancing weights is adjusted for every possible pitch of the crank pin axis, such that the resultant of the centrifugal forces exerted on the two weights passes through the point of intersection with the rotation axis of the perpendicular to the axis of the crank pin passing through the center of articulation of the crank pin on the wheel.

3. Device for balancing an articulated part around a center of oscillation bearing a crank pin centered on the axis passing through the center of gravity of the part and through the center of oscillation, said part articulating on a wheel driven rotationally around an axis passing through the center of oscillation and equipped with a means of varying the pitch of the part relative to the rotation axis by radial motion of the articulation of the crank pin on the drive wheel, wherein said device comprises two balancing weights rotationally integrated with the drive wheel, whose centers of gravity are placed on the side of the crank pin opposite the rotation axis in the plane passing through said axes, at different distances from the center of oscillation, and a means of controlling different radial spacings of the centers of gravity of the two weights from the rotation axis depending on the pitch of the axis.

4. Device for balancing an articulated part according to Claim 3, wherein each balancing weight is composed of two centrifugal weights rotationally integral with the drive wheel, whose centers of gravity are symmetrically movable by rotating around the rotation axis of a wheel on either side of the plane passing through the rotation axis and the axis of the crank pin, and wherein the means of controlling the radial spacings of the

balancing weights determines, for each balancing weight, the symmetrical rotations of the corresponding centrifugal weights around the rotation axis

5. Device for balancing an articulated part according to Claim 4, in which the articulation of the crank pin is mounted on a part that is radially slidable on the wheel, wherein the means of controlling the radial spacings of the center of gravity of each balancing weight comprises, for each centrifugal weight, a rack pivotably mounted on the slidable part that [words unintelligible due to typographical errors] with a first toothed pinion turning around an off-center axis on which a second pinion is centered and affixed, meshing with a toothed sector on the centrifugal weight and centered on the rotation axis, each centrifugal weight being guided along a circular guideway on the wheel and centered on the rotation axis.

[see original for Figures 1-6]

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**PATENT APPLICATION**

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**No. 75 03437**

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(54) Process and device for balancing a rotary part

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This invention concerns a process and a device for balancing a part that pivots around a center of oscillation, rotationally driven around an axis passing through the center of oscillation.

In particular, the invention is applicable to the balancing of swash plate mechanisms such as the one described in French Patent Application 73-39888 filed on November 9, 1973 by the same company.

The mechanism described in that patent application is especially applicable to an axial piston pump driven by means of a diagonal plate turning around a center of oscillation through the intermediary of a universal joint. The swash plate is rotationally driven by a crank pin running along the axis passing through the center of gravity of the plate and the center of oscillation, and pivoting around a wheel rotationally driven around the rotation axis.

The same patent application describes a means of controlling the output of the pump by the radial movement of the articulation of the crank pin on the drive wheel, with said movement controlling a variation in the pitch of the axis of the crank pin relative to the center of rotation axis.

In such pumps, the swash plate turns at a fairly high speed, and it is therefore necessary to balance it.

Dynamic balancing is difficult to achieve due to the output regulation, which varies the pitch of the axis of the plate.

The object of the invention is a new balancing process and a device to implement this process that enables balancing regardless of the pitch of the crank pin.

According to the invention, two radially movable balance weights are placed on the drive wheel, and their centers of gravity are placed on the side the axis of the crank pin opposite from the rotation axis, in the plane passing through said axes, and the different radial motions of the two balance weights are controlled according to the pitch of the axis of the crank pin relative to the rotation axis, such that the resultant of the centrifugal forces exerted on the two weights constitutes a counterweight whose variable intensity and position are capable of balancing the counterweight exerted by the rotary part for every possible pitch.

The invention will now be described with reference to a particular embodiment of a device for implementing the process according to the invention, provided as an example and illustrated in the attached drawings.

Figure 1 is a partial view of the drive mechanism of an axial piston pump.

Figure 2 and Figure 3 are diagrams illustrating the balancing process.

Figure 4 is a perspective view of the balancing device.

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Figure 6 is a cross section along VI-VI from Figure 5.

Figure 1 illustrates the drive mechanism of an axial piston pump comprising, inside a housing 1, a plurality of pistons 2 parallel to an axis 3, and evenly distributed around the latter. This figure shows a cross section of only one of the pistons 2. The pistons are connected by connecting rods 4, articulated at their ends to a diagonal plate 5, oscillatably mounted around a center of oscillation 6. In the illustrated embodiment, the plate 5 is swivelably mounted round an axis 51 on a cross-member 52, itself swivelably mounted on the housing 1, around trunnions 53 whose axes are perpendicular to the rotation axis 3 and the trunnion axis 51 of the plate. Since this assembly constitutes a universal joint, it is clear that the plate 5 is able to oscillate in all directions around the center 6, the point where the trunnion axes 51 and 53 meet.

Furthermore, on the side opposite the piston 1, the plate is equipped with an extension 54 that ends in a crank pin 55 whose axis passes through the center of oscillation 6 and is perpendicular to the trunnion axes 51 and 53. Preferably, the center of gravity of the assembly is placed on the axis 550 of the crank pin.

The crank pin is connected, through the intermediary of a ball and socket joint 57, to a wheel 7 centered on the rotation axis 3 and rotationally driven around said axis. To liberate the central part of the wheel, it is guided, along its circumference, in the end 10 of the housing 1, by two tapered roller bearings 71.

As shown in Figures 4 and 5, the spherical bearing 57 of the crank pin is mounted on a sliding nut 72 guided in a guideway 73 in the central part of the wheel 7, whose axis passes through the center of the wheel 7 located on the rotation axis 3.

As indicated in the aforementioned previous application, the assembly that has just been described makes it possible to easily vary the output of the pump while it is operating, since the pitch of the piston thrust plate 5, and thus its travel, can be adjusted.

Of course, all mobile parts are balanced, to the extent possible, and, thus, the center of gravity of the oscillating plate is normally on the axis 550 of the crank pin. Balancing the drive wheel 7 is somewhat more difficult, inasmuch as the piston that controls the travel of the sliding nut has a radially variable position.

Therefore, it may be necessary to take the counterweighting of the wheel itself into account. However, the greatest counterweighting clearly comes from the reaction of the crank pin on the drive wheel. Figure 2 schematically illustrates the rotation axis 3, the crank pin 55 and its axis 550 passing through the center of oscillation 0, on which the center of gravity G of the oscillating plate with a mass M is located. If d is the distance OG from the center of gravity to the center of oscillation, the counterweighting from the rotation of the oscillating plate is equal to  $M_d \sin A$ .

The balance equations show that the balancing counterweight should be in the plane  $P_e$  perpendicular to the rotation axis 3, passing through the point of intersection D of the axis 3, with the perpendicular to the axis 550 raised from the center of articulation of the crank pin 55 on the wheel 7. A turning weight is generally balanced with two counterweights traveling in two balancing planes. One of the characteristics of the invention consists precisely of using only one balancing plane, by positioning the center of gravity G of the oscillating plane such that the dynamic reactions at the center of oscillation O are nil.

We see that, if we use a balancing weight with a value  $M_e$ , it should be placed in the balancing plane and in the plane passing through the axis 3 and the axis 550, at a radial distance  $R_e$  from the axis 3, such that:

$$M_e R_e = M_d \sin A$$

It is clear that the counterweighting depends on the pitch A of the oscillating plate. Since the balancing weight  $M_e$  cannot be varied, its position is varied such that the radius of turn of the center of gravity is equal to:

$$R_e = \frac{M_d \sin A}{M_e}$$

However, in view of the foregoing, we see that it is also necessary to vary the position of the balancing plane  $P_e$ , i.e., the distance e between the plane  $P_e$  and the plane of the wheel 7.

According to the invention, the variation of the balancing plane is achieved by using two balancing weights placed at different radial spacings from the axis 3.

For greater convenience, each balancing weight is composed of two centrifugal weights affixed so as to turn symmetrically around the axis 3 such that their center of gravity remains at a constant distance from the axis. In that way, as illustrated in Figure 3, if we call  $g_1$  and  $g_2$  the centers of gravity of the centrifugal weights, with  $m_1$  and  $m_2$  arc with thus describing an/ circle with a radius r around the center of the wheel, we see

that the two centrifugal weights  $m_1$  and  $m_2$  are equivalent to a single weight with a value of  $2m$  placed at a distance R from the center of rotation equal to the projection of r on the axis of symmetry.

Thus, by causing the two centrifugal weights to turn symmetrically around the center of the wheel, we can produce a counterweight with a variable value in the plane perpendicular to the rotation axis, passing through the centers of gravity of the two weights.

According to one characteristic of the invention, to produce a counterweight not only with a variable value, but in a balancing plane with a variable position along the axis, we use two counterweights placed on either side of the plane of the wheel, which we

call B and B'. As illustrated in Figure 2, the two counterweights B and B' are placed in balancing planes P and P', one of which, P, is located between the center of oscillation O and the plane P<sub>r</sub> of the wheel, and the other, P', on the other side of the plane P<sub>r</sub> of the wheel. The position of the balancing planes P and P' depends on the conditions under which the centrifugal weights may be placed on the wheel. However, the plane P' must be placed at a distance from the plane P<sub>r</sub> of the wheel that is greater than the maximum value of the distance e between the plane P<sub>r</sub> of the wheel and the theoretical balancing plane P<sub>e</sub> for the maximum pitch of the oscillating plate.

By varying the values of the counterweights B and B', it is possible to arrange for the resultant to be located in the theoretical balancing plane P<sub>e</sub>.

Figures 4, 5, and 6 illustrate, as an example, a device that provides this result.

We have already indicated that the wheel 7 is guided and driven by its periphery, which makes it possible to entirely liberate the central part where the sliding nut 72 supporting the hinge of the crank pin and the travel cylinder 73 of the sliding nut are essentially located.

The wheel 7 is equipped, in its equatorial plane, with a disc 75 in whose central part a radial opening 750 is pierced, within which the sliding nut can travel. The guides 73 of the sliding nut are placed along the sides of the opening 750, parallel to the travel axis 720 of the sliding nut. On either side of the baffle 75, there are two pairs of centrifugal weights 8 and 8'. Each pair is composed of two centrifugal weights symmetrically guided on either side of the axis 720 along circular guideways centered on the rotation axis 3. As illustrated in greater detail in Figure 6, each centrifugal weight 8 is guided between an inner guideway 76 in the shape of a sector of a circle, which is integral with the corresponding guideway 73 of the sliding nut, and an outer guideway 77 affixed to a bead that is integral with the periphery of the wheel 7. Each centrifugal weight 8 is equipped with fillets that adapt to the two guideways and rest also on a circular bead 741 on the wall 75. Thus, each centrifugal weight may be moved along the guideways 77 and 76, with its center of gravity remaining at a constant distance from the axis 3 in a plane perpendicular to said axis.

The circular guideways are placed such that the centrifugal weights may come as close as possible to the axis 720, as illustrated at the right in Figure 5, and may move away from said axis until their center of gravity comes into the radial plane perpendicular to the axis 720, in the position illustrated at the left in Figure 5. Thus, we see that the counterweight resulting from the action of the two centrifugal weights varies between a nil value and a maximum value. Depending on the shape and mass of the centrifugal weights, we may choose the balancing planes P and P' and the maximal values of the counterweights.

Figure 2 illustrates the planes P and P' in which the counterweights B and B' are produced. These are located, respectively, at distances a and b from the plane of the wheel in which the counterbalance  $B_r$  of the wheel itself may be located. If we take as a reference the axis Oz perpendicular to the rotation axis 3, passing through the center of articulation of the crank pin, the equation for the projection on the axis Oz gives:

$$(3) B' + B - B_e + B_r$$

$B_e$  being the balancing counterweight that must be produced in the theoretical plane  $P_e$ .

The equation for the momentum relative to the center O gives:

$$(4) a B' - b B = e B_e$$

From this, we derive the values that must be used for the counterweight produced by the centrifugal weights:

$$(5) B' = \frac{(e+b) B_e + b B_r}{a+b}$$

and:  $(6) B = B_e + B_r - B'$

Furthermore, if we call l the distance OC between the center of oscillation and the center of the wheel, we may write:

$$(7) e = l \tan^2 A$$

For any pitch value A of the oscillating plate, the relationship (1) gives  $B_e = M_e R_e = M_d \sin A$ . If we determine in advance the distances a and b of the balancing planes P and P' relative to the plane of the wheel, relationships 5 and 6 make it possible to calculate the outer and inner counterweights B' and B.

If m and m' are the masses of the balancing centrifugal weights, r and r' the radii of turn of the centers of gravity of the centrifugal weights, R and R' the projections on the axis 720 of the positions of the centers of the centrifugal weights, and L the pitch relative to the axis 720 of the radii passing through the centers of gravity, where:

$$B = 2m R = 2m r \cos L$$

and:  $B' = 2m' R' = 2m' r' \cos L$ ,

it is easy to deduce from the previously calculated values of the counterweights B and B' the angle L as a function of the angle A.

Then, it is sufficient to develop a mechanical system whereby the angular position of the centrifugal weights is linked to the radial position of the sliding nut 72 such that the desired value L corresponds to each value of A. Then, there is only the problem of choosing a mechanism, which can be determined more or less empirically.

We have found that the mechanical device illustrated as an example in the figures made it possible to achieve this variation in the pitch of the centrifugal weights.

As illustrated in Figures 4, 5, and 6, for each pair of centrifugal weights, this mechanical device is essentially composed of a pair of racks 91, articulated around the sliding nut around axes parallel to the rotation axis, running symmetrically along either



side of the axis 720. Each rack is affixed, by means of a U bolt 92 that meshes with a pinion 93 rotationally mounted on the wall 75 around an off-center pivot 94. The racks 92 themselves are rotationally integral with the off-center pivots 93.

On each pivot 94 is centered a pinion 95 that meshes with a toothed sector that is integral with the centrifugal weight 8 and centered on the rotation axis 3.

Thus, the travel of the sliding nut 72 by means of the cylinder 73 drives, through the intermediary of the racks 91, the rotation of the off-center pivots 93 and as a result the pivots 95, which in turn control the rotation of the centrifugal weights around the axis 3, since their pivots 94 are stationary relative to the wheel. In view of the off-centering of the pinions 93 and the gear ratio between the pinions 95 and the sectors 85, we see that the travel of the sliding nut, i.e., the variation of the angle  $A$ , results in an irregular variation in the angle  $L$ .

We see that it is possible to set up equations for everything and, by varying certain parameters within the limits allowed by the mechanical construction considerations (dimensions, strength, etc.), we can find, using a computer for example, the optimal solution for best following the theoretical variation in  $L$  as a function of  $A$ , established as indicated above.

Of course, to simplify the calculations, it is possible to not take into account the counterweighting of the wheel due to the variation in the control piston of the sliding nut, and to balance the wheel once and for all.

Conversely, to achieve the most accurate results, it is possible to take into account the counterweighting of the wheel due to the non-colinear motion of the universal joint through the intermediary of which the plate is oscillatingly mounted, which requires secondary counterweights. However, this effect may be generally considered negligible relative to the primary counterweight.

Furthermore, the mechanical system for varying the angle  $L$  as a function of the position of the crank pin is provided solely for information and it is possible to imagine other systems that are not necessarily mechanical, but rather hydraulic, for example.

### CLAIMS

1. Process for balancing a rotary part articulated around a center of oscillation and rotationally driven around an axis passing through the center of oscillation through the intermediary of a crank pin running along the axis passing through the center of gravity of the part and the center of oscillation and connected to a drive wheel turning around the rotation axis, through the intermediary of a radially movable articulation on the wheel controlling the variation of the pitch of the axis of the crank pin relative to the rotation axis, wherein, placed on the drive wheel, are two radially movable balancing weights whose centers of gravity are placed on the side of the axis of the crank pin opposite from the rotation axis, in the plan passing through said axes, and wherein the different radial spacings of the two balancing weights are determined by the pitch of the axis of the crank pin relative to the rotation axis, such that the resultant of the centrifugal forces exerted by the two weights constitutes a counterweight with variable intensity and position, capable of balancing the counterweighting exerted by the turning part for every possible pitch.

2. Balancing process according to Claim 1, wherein the position of the centers of gravity of the balancing weights is adjusted for every possible pitch of the crank pin axis, such that the resultant of the centrifugal forces exerted on the two weights passes through the point of intersection with the rotation axis of the perpendicular to the axis of the crank pin passing through the center of articulation of the crank pin on the wheel.

3. Device for balancing an articulated part around a center of oscillation bearing a crank pin centered on the axis passing through the center of gravity of the part and through the center of oscillation, said part articulating on a wheel driven rotationally around an axis passing through the center of oscillation and equipped with a means of varying the pitch of the part relative to the rotation axis by radial motion of the articulation of the crank pin on the drive wheel, wherein said device comprises two balancing weights rotationally integrated with the drive wheel, whose centers of gravity are placed on the side of the crank pin opposite the rotation axis in the plane passing through said axes, at different distances from the center of oscillation, and a means of controlling different radial spacings of the centers of gravity of the two weights from the rotation axis depending on the pitch of the axis.

4. Device for balancing an articulated part according to Claim 3, wherein each balancing weight is composed of two centrifugal weights rotationally integral with the drive wheel, whose centers of gravity are symmetrically movable by rotating around the rotation axis of a wheel on either side of the plane passing through the rotation axis and the axis of the crank pin, and wherein the means of controlling the radial spacings of the

balancing weights determines, for each balancing weight, the symmetrical rotations of the corresponding centrifugal weights around the rotation axis

5. Device for balancing an articulated part according to Claim 4, in which the articulation of the crank pin is mounted on a part that is radially slidable on the wheel, wherein the means of controlling the radial spacings of the center of gravity of each balancing weight comprises, for each centrifugal weight, a rack pivotably mounted on the slidable part that [words unintelligible due to typographical errors] with a first toothed pinion turning around an off-center axis on which a second pinion is centered and affixed, meshing with a toothed sector on the centrifugal weight and centered on the rotation axis, each centrifugal weight being guided along a circular guideway on the wheel and centered on the rotation axis.

[see original for Figures 1-6]